CFD ANALYSIS FOR NATURAL CONVECTION OF A VERTICAL TUBE WITH VARIOUS FIN CONFIGURATIONS

A THESIS SUBMITTED IN THE PARTIAL FULFILLMENT OF THE REQUIREMENT FOR THR DEGREE OF

BACHELOR OF TECHNOLOGY
IN
MECHANICAL ENGINEERING
By

ASHISHMAN KAR

DEPARTMENT OF MECHANICAL ENGINEERING
NATIONAL INSTITUTE OF TECHNOLOGY
ROURKELA-769008
CLASS OF 2008~2012

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UNDER THE GUIDANCE OF

Dr. ASHOK KUMAR SATAPATHY

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CERTIFICATE

This is to certify that the thesis entitled as "CFD ANALYSIS FOR NATURAL CONVECTION OF A VERTICAL TUBE WITH VARIOUS FIN CONFIGURATIONS" submitted by Sri ASHISHMAN KAR in partial fulfillment of the requirements for the award of Bachelor of technology Degree in Mechanical Engineering at the National Institute of Technology, Rourkela is an authentic work carried out by him under my supervision and guidance.

To the best of my knowledge, the matter embodied in the thesis has not been submitted to any other University/ Institute for the award of any degree or diploma.

DATE:

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TABLE OF CONTENTS:

LIST OF FIGURES:

LIST OF TABLES

CHAPTER 1

Introduction

1.1 ABSTRACT:

This project is about a thorough study of natural convection from a heated pipe having fins of various configurations using ANSYS WORKBENCH version 13.0. The material under consideration is aluminium and the free stream fluid is air. The heat transfer rate from the fins, outer wall and the overall heat transfer rate has been calculated and compared for various fin configurations. Also the surface nusselt number and surface overall heat transfer co-efficient has been found out. Temperature contours for various fin configuration has been plotted showing the convection loops formed around the heated pipe surface. Velocity contours for various fin configurations has been plotted and the motion of heated fluid is shown. Plots for nusselt number and heat transfer co-efficient are also shown.

The assumptions during the analysis have been taken considering the manufacturing and practical applications and working conditions. Hence the results obtained can be referred to while solving any such kind of problems in the practical field where only natural convection is under consideration.

After comparing it is shown that we can find that the best configuration for this type of convective heat transfer of a heated pipe is a TRAPEZOIDAL fin as they have the highest total heat transfer rate

1.2 GENERAL INTRODUCTION

Heat exchangers are widely used in various, transportation, industrial, or domestic applications such as thermal power plants, means of heating, transporting and air conditioning systems, electronic equipment and space vehicles. In all these applications improvement in the efficiency of the heat exchangers can lead to substantial cost, space and material savings. Hence considerable research work has been done in the past to seek effective ways to improve the efficiency of heat exchangers. The referred investigation includes the selection of fluid with high effective heat transfer surfaces made out of high conductivity materials, high thermal conductivity and selection of their flow arrangements. For both single and two phase heat transfer effective heat transfer enhancement techniques have been reported. However in the present work only SINGLE PHASE STEADY STATE NATURAL CONVECTION technique has been considered. The heat transfer enhancement methods reported in publications be summarized in many forms but primarily they may be grouped as active enhancement methods.

The basis of any heat transfer enhancement technique lies in the utilization of some external power in order to permit the mixing of working fluids, the rotation of heat transfer surfaces, the vibration of heat transfer surfaces or of the working fluids also the generation of electrostatic fields.

The major heat transfer enhancement techniques that have found widely spread commercial application are those which possess heat transfer enhancement elements. All passive techniques aim for the same, namely to achieve higher values of product of the heat transfer coefficient and heat transfer surface area. A distinguish between the way how the heat transfer enhancement is achieved, is common in the heat transfer community. Here in the present work, a terminology similar to the literature is followed although for practical applications are irrelevant how the heat transfer enhancement is achieved.

The choices of the particular passive method depend greatly on the mode of the convective heat transfer (natural or forced convection) and on the fluids used to transfer heat. When argumentation of heat transfer has to be provided, the thermal resistance in the direction of the heat flow has to be considered. E.g. it is not advantageous to invest in the reduction of already low thermal resistance. It is known that gases, owing to their low thermal conductivity, are characterized with much higher resistance for the heat flow compared with liquids. Therefore in gas-liquid heat exchangers, the argumentation measures should generally be applied to the gas side.

The most effective heat transfer enhancement can be achieved by using fins as elements for the heat transfer surface area extension. In the past a large variety of fins have been applied for these purposes, leading a very compact heat exchangers with only gas or gas and liquid as the working media. Plate fin rotary regenerators and tube fin are widely encountered compact heat exchangers across the industry. Here the area of interest is the tube fin configuration. These are built as a combination of tubes with various cross sections with fins present both outside and inside the tubes. The common form of the tube cross-section is round or rectangular, but elliptical cross-sections are also encountered. Fins are generally attached by means of tight mechanical fit, adhesive bonding, soldering, brazing, and welding or by extrusion. Depending upon the form and direction of the fins, the tubes may be classified as individual tube with normal fins, individual tubes with longitudinal fins or tube arrays with plain, wavy or interrupted external of internal fins.

1.3 AIM OF THE WORK

There are almost no industrial fields in which heat exchangers are not applied. The design of the heat exchangers influence greatly the design of the entire system or process in which they are applied. Many factors influence the design of a heat exchanger, but the most important one is the heat transfer rate. With an exception of a few cases usually high heat transfer rate and small pressure drop in a small volume is needed in all kind of usual processes.

Heat is generally transferred in three basic forms: conduction, convection, radiation. The intensity of heat conduction is not a challenging problem and usually can be controlled by material chosen to build the system. Further radiation is of very less concern when the heat transfer process happens in moderate temperatures. The intensity of heat transferred by convection is the dominant aspect in this kind of analysis as compared to that by conduction and radiation.

Based on Newton's law of cooling, convective heat transfer can be calculated as the product of heat transfer coefficient, heat transfer surface area and the temperature difference between the wall of the tube and the fluid flowing inside the walls. The wall to fluid temperature difference is usually adjusted oneself based on the operating conditions and therefore it cannot be used to enhance the heat transfer rate. One can increase the heat transfer surface area or the heat transfer coefficient, or both of them simultaneously. But as the heat transfer coefficient for a specific material at specific temperature is constant, hence the only way of changing the heat transfer rate is to vary the heat transfer surface area.

Interrupted fins in the form of strips or louvered fins provide both a heat transfer surface area increase and also increase in the Effective heat transfer coefficient. Therefore these are particularly effective in obtaining high heat transfer rates. The mechanism which leads to high heat transfer coefficients of such fins is the periodic interruption of boundary layer around the fins and in this way also achieving better mixing with different temperature fluid streams.

The exchange of heat energy is studied on a tube with circular cross-section and with specific inner and outer radius having outer disc shaped fins. The fins attached with the tube can be of variable shape and size. Three basic types of fins are considered and the transfer of heat energy from a tube with such fin configurations is estimated.

The design calculations of the tube and the fin dimensions are done based upon equations suitable for the maximum heat transfer rate at low production costs. The material used for the calculations is considered to be ALUMINIUM. Both the tube and fins are considered to be made up of Aluminium and the fluid inside the tube is Water.

ANSYS 13.0 WORKBENCH version is used for the entire simulation processes. Experimental values of the working temperatures and corresponding properties for the fin and tube material along with water is considered and fed to the software.

The convection type under consideration is NATURAL CONVECTION. The tube is vertically situated and vertical flow is considered for calculation. A very minimal fluid velocity is assumed and the entire heat transfer process is made to happen under the influence of gravity.

The objective of the last part of the project is to plot various contours suggesting the ease of heat transfer with various fin cross-sections for example temperature contours, velocity contours across the length and cross section of the pipe. Various graphs suggesting the heat transfer rates such as nusselt no plot is also drawn by the software. The final objective of the project is to compare the results and to find out the best fin cross-section for the specified working conditions. Also the results are compared with that of different fin configurations (external and internal spiral fins) to find out the best fin configuration for the working conditions.

CHAPTER 2

Preliminary considerations on heat transfer and enhancement techniques.

2.1 BASIC HEAT TRANSFER:

2.1.1 HEAT TRANSFER AND THERMODYNAMICS:

The study of transfer phenomenon which includes transfer of momentum, energy, mass etc has been recognized as a unified discipline of fundamental importance on the basis of thermodynamic fluxes and forces. The transfer of such phenomena occurs due to a conjugate force of temperature gradient, velocity gradient, concentration gradient chemical affinity etc. The transfer of heat energy due to temperature difference or gradient is called heat transfer.

2.1.2 MODES OF HEAT TRANSFER:

The modes of heat transfer can be divided into three segments.

- CONDUCTION
- RADIATION
- CONVECTION

2.1.2.1 CONDUCTION:

CONDUCTION refers to the transfer of heat between two bodies or two parts of the same body through molecules which are, more or less, stationary, as in the case of solids.

The governing equation for conductive heat transfer is:

In **Cartesian coordinates**

$$\Delta f = \frac{\partial^2 f}{\partial x^2} + \frac{\partial^2 f}{\partial y^2} + \frac{\partial^2 f}{\partial z^2} = 0.$$

In **cylindrical coordinates**,

$$\Delta f = \frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial f}{\partial r}\right) + \frac{1}{r^2}\frac{\partial^2 f}{\partial \phi^2} + \frac{\partial^2 f}{\partial z^2} = 0$$

In **spherical coordinates**,

$$\Delta f = \frac{1}{\rho^2}\frac{\partial}{\partial \rho}\left(\rho^2\frac{\partial f}{\partial \rho}\right) + \frac{1}{\rho^2 \sin\theta}\frac{\partial}{\partial \theta}\left(\sin\theta\frac{\partial f}{\partial \theta}\right) + \frac{1}{\rho^2 \sin^2\theta}\frac{\partial^2 f}{\partial \varphi^2} = 0.$$

2.1.2.2 RADIATION:

Thermal radiation refers to the radiant energy emitted by bodies by virtue of their own temperatures, resulting from the thermal excitation of the molecules. Radiation is assumed to propagate in the form of electromagnetic waves.

The governing equation for Radiation heat transfer is:

PLANK'S LAW:

$$B_\nu(T) = \frac{2h\nu^3}{c^2}\frac{1}{e^{\frac{h\nu}{k_B T}} - 1}, \text{ or } B_\lambda(T) = \frac{2hc^2}{\lambda^5}\frac{1}{e^{\frac{hc}{\lambda k_B T}} - 1}$$

2.1.2.3: CONVECTION

When energy transfer takes place between a solid and fluid system in motion, the process is known as *convection.* If the fluid motion is impressed by compressor or pum, it is called FORCED CONVECTION. If fluid motion is caused due to density difference, it is called NATURAL CONVECTION.

2.2 TYPES OF CONVECTION:

2.2.1 NATURAL CONVECTION:

Natural convection is a mechanism, or type of heat transfer in which fluid motion is not generated by any external source like pump, fans, suction devices etc but only due to density difference in the fluids occurring due to temperature gradient.

The driving force of natural convection is Buoyancy, a result of difference in fluid density. Because of this the presence of a proper acceleration which would provide sufficient resistance to gravity or an equivalent force is essential for natural convection.

2.2.2 FORCED CONVECTION:

Forced convection is mechanism, or type of heat transfer in which fluid motion is generated by an external source like pump, fans, suction devices etc. it is considered as the main method of useful heat transfer as significant amount of heat energy can be transferred by this process.In forced convection cases some amount of natural convection is always present. This type of convections is called as MIXED CONVECTION.

2.2.3 GRAVITATIONAL OR BYUOYANT CONVECTION:

This type of convection (a type of NATURAL CONVECTION) is induced by natural buoyancy variation resulting from material properties other than temperature. Typically this is caused by variable composition of fluid, or concentration gradient (SOLUTAL CONVECTION)

2.2.4 THERMO-MAGNETIC CONVECTION

Thermo magnetic convection can occur when an external magnetic field is imposed on a Ferro fluid of varying magnetic susceptibility. In the presence of temperature gradient this results in a non-uniform magnetic body force which leads to fluid movement. (Ferro fluid: liquid which becomes strongly magnetized when subjected to magnetic field)

This type of convection is useful in miniature micro-scale devices

2.3 NATURAL CONVECTION:

The measures of Natural convection are:

2.3.1 REYNOLDS NUMBER (R_e):

The Reynolds Number, the non-dimensional velocity, is defined by the ratio of dynamic pressure (ρ u2) and shearing stress (μ u / L)

$$\boxed{R_e = \rho u L / \mu}$$

2.3.2 NUSSELT NUMBER (N_u):

In heat transfer at a boundary (surface) within a fluid, the Nusselt number is the ratio of convective to conductive heat transfer across (normal to) the boundary. In this context, convection includes both advection and conduction. It is a dimensionless number. The conductive component is measured under the same conditions as the heat convection but with a (hypothetically) stagnant (or motionless) fluid.

$$\boxed{\mathrm{Nu}_L = \frac{hL}{k_f}}$$

2.3.3 PRANDTL'S NUMBER (P_r):

The Prandtl number is a dimensionless number; the ratio of momentum diffusivity (kinematic viscosity) to thermal diffusivity.

$$P_r = V / \alpha$$

2.3.4 GRASHOF'S NUMBER (G_r):

The Grashof number is a dimensionless number in fluid dynamics and heat transfer which approximates the ratio of the buoyancy to viscous force acting on a fluid. It frequently arises in the study of situations involving natural convection.

$$\mathrm{Gr}_L = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2}$$ for vertical flat plates

$$\mathrm{Gr}_D = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2}$$ for pipes

$$\mathrm{Gr}_D = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2}$$ for bluff bodies

2.3.5 RAYLEIGH'S NUMBER (R_a):

is defined as the product of the Grashof number, which describes the relationship between buoyancy and viscosity within a fluid, and the Prandtl number, which describes the relationship between momentum diffusivity and thermal diffusivity. Hence the Rayleigh number itself may also be viewed as the ratio of buoyancy and viscosity forces times the ratio of momentum and thermal diffusivities.

$$\mathrm{Ra}_x = \mathrm{Gr}_x \mathrm{Pr} = \frac{g\beta}{\nu\alpha}(T_s - T_\infty)x^3$$

2.4 HEAT TRANSFER BY EXTENDED SURFACE:

Convection heat transfer is governed by the relation:

$$Q = h A (T_w - T_\infty)$$

To increase the heat transfer rate the following ways can be adopted.

1. Increasing heat transfer co-efficient (h). However increasing the value of h does not significantly influence the value of Q.
2. Surrounding fluid temperature (T_∞) can be decreased. But it is often impractical as in most cases the surrounding is atmosphere.
3. Hence the only way is by increasing the surface area across which convection occurs.

The increase in cross sectional convection area can be achieved by using fins that exted from the wall of the convection shell. The thermal conductivity of the fin material has a very strong effect on the temperature distribution across the wall of the convection shell and thus the degree to which the heat transfer rate is enhanced.

Various types of fins are usually used:

- Straight fins of uniform cross section
- Straight fins of non-uniform cross section
- Annular fins
- Cylindrical fins
- Pin fins

2.5 FIN PERFORMANCE:

2.5.1 FIN EFFECTIVENESS:

Fin effectiveness is defined as the ratio between heat transfer rate with fin and heat transfer rate without fin.

$€_f = Q_o / hA\ \theta_o$

while using a fin for increasing heat transfer rate we should consider that, the fin itself represents a conductive resistance to heat transfer from original surface. Therefore it is not necessary that by using fins the heat transfer rate increases.

This facto is calculated by fin effectiveness

When $€_f < 2$, the use of such fins are not justified.

Fin effectiveness can be enhanced by,

1. Choice of material of high thermal conductivity. Eg. Aluminium, Copper
2. Increasing ratio of area to the perimeter of the fins. The use of thin closely placed fins is more suitable than thick fins.
3. Low values of heat transfer coefficient (h).

2.5.2 FIN EFFICIENCY:

This is the ratio of the fin heat transfer rate to the heat transfer rate of the fin if the entire fin were at the base temperature.

$$\boxed{\eta_f = \frac{q_f}{hA_f\theta_b}}$$

CHAPTER 3

DESIGN CALCULATIONS:

3.1 HEAT TRANSFER BY CYLINDRICAL OR DISC SHAPED FINS:

The area normal to the heat flux vector can be written as

$$A = 2 \pi r b$$

And the peripheri can be expressed as: $P = 4 \pi r b$

For the annular element of radius r and thickness dr: By energy balancing:

$$-k * 2 \pi r b * dT/dr =$$

$$-k * 2 \pi (r + dr) b (dT/dr + d^2T/dr^2 * dr) + 4 h \pi r dr$$

Or, $d^2T/dr^2 + dT/rdr – 2 h/k b(T-T_\infty) = 0$

Let $\theta = T - T_\infty$

Or, $d^2\theta / dr^2 + d\theta / rdr – (2 h / kb) \theta = 0$

This is BESSEL’s equation of zero order.

Solution is,

$$\Theta = C_1 I_0 (mr) + C_2 K_0 (mr)$$

Where, $m = (2 h / k b)^{1/2}$

I_0 = modified Bessel’s function 1st kind zero order.

K_0 = modified Bessel’s function 2nd kind zero order.

C_1 & C_2 are arbitary constants.

At $r = r1$, $T = T_w$, $\theta = Tw - T\infty$

And at $r = r2$, $dT/dr = 0$ or $d\theta / dr = 0$ since $b << (r2 – r1)$

Using these relations,

$$\theta/\theta_0 = \frac{I0\,(mr)\;\; K1\,(mr2)\; +\; K0\;\, (mr)\;\, I1\;\, (mr2)}{I0\,(mr1)\;\; K1\,(mr2)\; +\; K0\;\, (mr1)\;\, I1\;\, (mr2)}$$

where,

$I_1(mr)$ = bessel's function of order 1 1^{st} kind

$K_2(mr)$ = bessel's function of order 1 2^{nd} kind

Fin efficiency is given by,

$$Q_0 \;/\; 2\,\pi\, h\,(r2^2 - r1^2)\,\theta_0$$

Or, $2\, r1 \,/\, m(r2^2 - r1^2)$ x

$$k1\,(mr1)\, I_1\,(mr2) - I1\,(mr1)\, K1\,(mr2) \,/ K_0\,(mr1)\, I_1\,(mr2) + I0\,(mr1)\, K1\,(mr2)$$

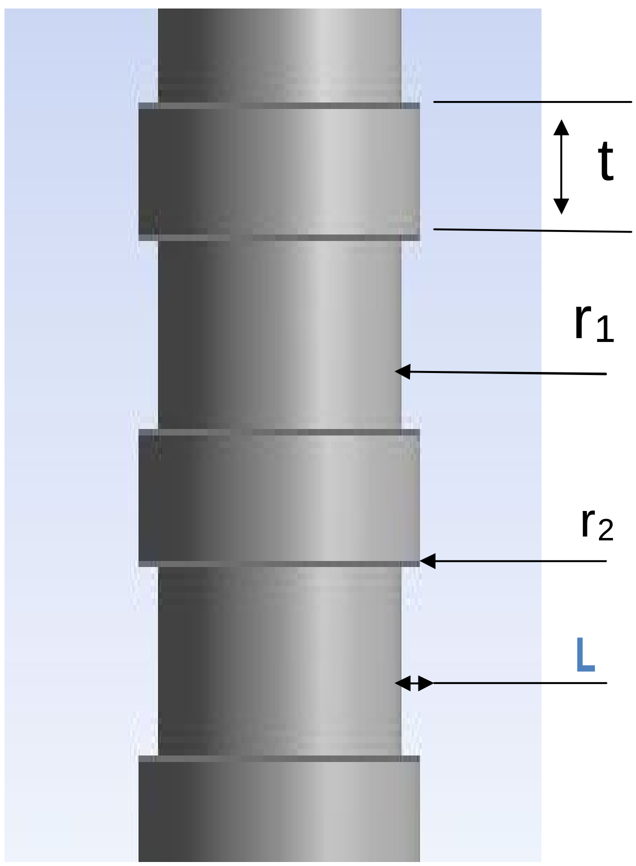


FIGURE1

3.2 LIMITATIONS OF EXTENDED SURFACES:

The installation of fins on a heat transfering surface increases the heat transfer area but it is not necessary that the rate of heat transfer would increase.

For long fins the rate of heat loss from the fin is,

$$Q = (h\,p\,K\,A)^{1/2} = K\,A\,(h\,p / K\,A)^{1/2}\ \theta_0 = KAm\,\theta_0$$

When $h / mK = 1$ or $h = mK$ hence $Q = hA\,\theta_0$

Which is same as the rate of heat transfer without fins. Thus when $h = m\,K$ the extended surface doesnot increase rate of heat transfer what ever its area may be.

$$\text{For } h / m\,K > 1\ ,\ Q < h\ A\ \theta_0$$

Which is less than the heat transfer rate without fins,Hence fins reduces the heat transfer.

Hence fins are used only when h / k is very low for a given geometry where the heat transfer would be more efficient

3.3 FINS OF MINIMUM WEIGHT:

For designing of cooling devices on vehicles, especially air crafts, the problem for exchanging maximum amount of heat with the least weight addition arises.Reducing the weight also reduces the cost of the finned heat exchanger mechanism. For a rectangular annular fin the fin cross section should be rectangular,

$$\text{So, } m = (2\,h / k\,b)^{1/2}$$

When the tip loss is neglected,

$$Q_0 = (h\,p\,K\,A)^{1/2}\,\theta_0\,\tanh mL = m\,K\,A\,\theta_0 \tanh m\,L$$

$$= (2\,h / K\,b)^{1/2}\,K\,b\,L\,\theta_0 \tanh((2\,h / K\,b)^{1/2}\,A_1 / b)$$

$$= (2\,h\,K)^{1/2}\,b^{½}\,L\,\theta_0\,\tanh(2\,h / K)^{1/2}\,(A_1 / b^{3/2})$$

$$dQ_1/db = (2\,h\,K)^{1/2}\,L\,\theta_0\,\tanh((2\,h / K)^{1/2}\,(A_1 / b^{3/2}))b^{-1/2}/2 \; +$$

$$(2\,h\,K)^{1/2}\,L\,\theta_0 b^{1/2} / [\cos h^2((2\,h / K)^{1/2}(A_1/b^{3/2}))(2h/K)^{1/2}A_1(b)^{5/2}]$$

$$= 0$$

Solving the above equation we have,

$$l/(b/2) = 1.419\ (2k/hb)^{1/2}$$

3.4 BOUSSINESQ APPROXIMATION

Equations governing natural convection:

CONTINUITY EQUATION : $\delta u/\delta x + \delta v/\delta x = 0$

MOMENTUM EQUATION : $\rho\ C_P\,(u\,\delta u/\delta x + v\,\delta u/\delta y) =$

$$-\rho g - \delta p/\delta x + \mu\,\delta^2 u/\delta y^2$$

ENERGY EQUATION : $\rho\ C_P\,(u\,\delta T/\delta x + v\,\delta T/\delta y) = K\,\delta^2 T/\delta y^2$

In natural convection the term – ρg on the right hand side of the momentum equation represents the body force exerted on the fluid element in negative x direction. For the small temperature differences the density ρ in the buoyancy term is considered to vary with temperature whereas ρ appearing elsewhere is constant. This is called Boussinesq approximation.

ANALYSIS

4.1 ANALYSIS FOR TUBE WITHOUT FINS:

4.1.1 MESS STATISTICS AND PARAMETERS ASSUMED DURING ANALYSIS:

PHYSICAL PARAMETERS	VALUES
Type of fin	Without fins
Cross section of the tube	Circular
External diameter of the pipe	50 mm
Length of the pipe	150 mm
Free stream fluid	Air
Material for tube and fins	Aluminium
Model for convection	Bousinessq
Tube wall temperature	380 k
Free stream air temperature	300 k
Convection heat transfer coff	10 W/m^{2}k
MESH PARAMETERS	VALUES
Messing method	Trapezoid
Relevance sizing centre	Fine
Element size	0.0001m
Initial size seed	Active assembly
Smoothing	High
Transition	Slow
Span angle centre	Fine
Number of nodes	1794
Number of elements	1474
Orthogonality quality	7.32e-01
Aspect ratio	1.55e01

TABLE 1

4.1.2 TEMPERATURE CONTOUR:

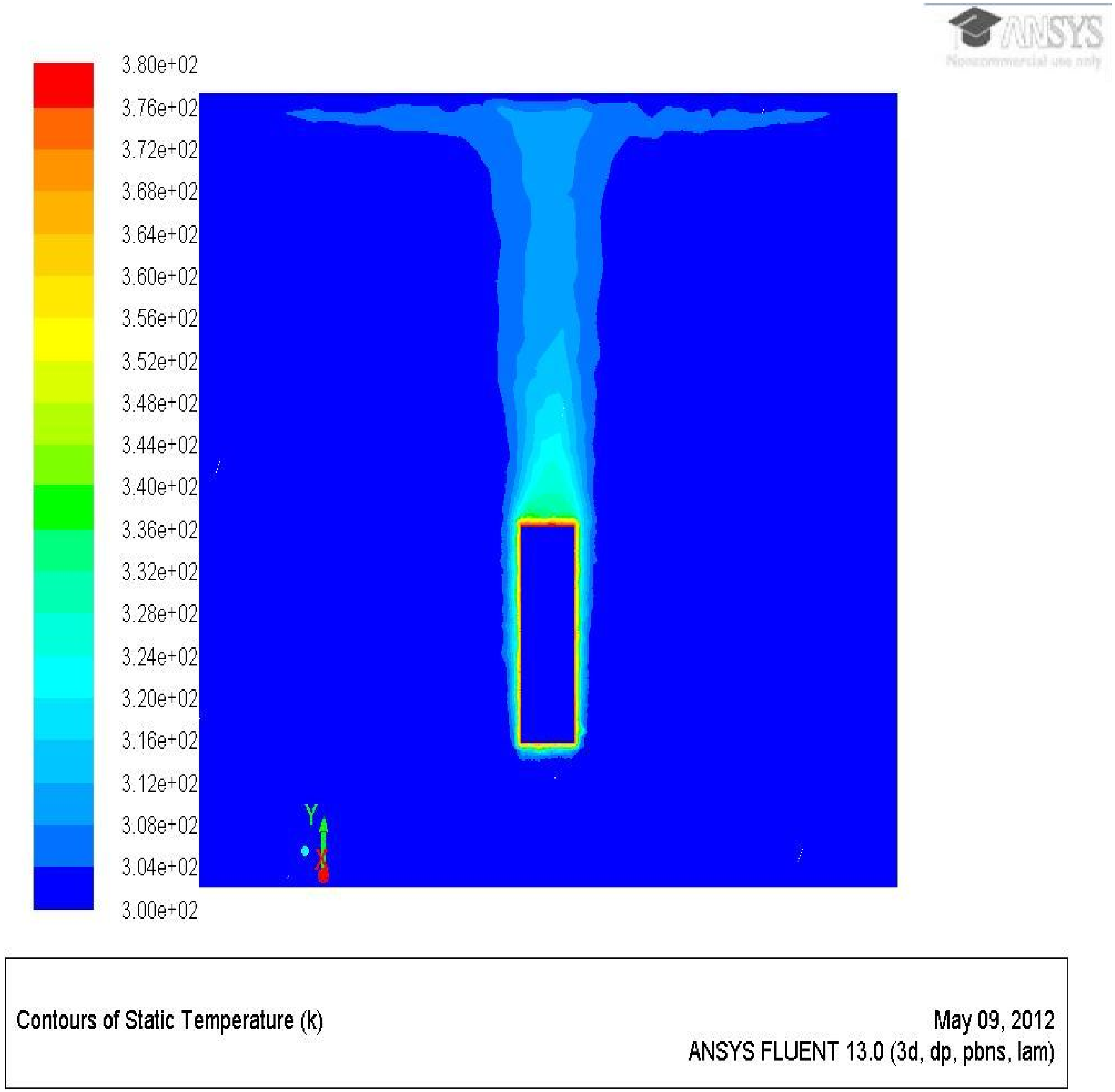
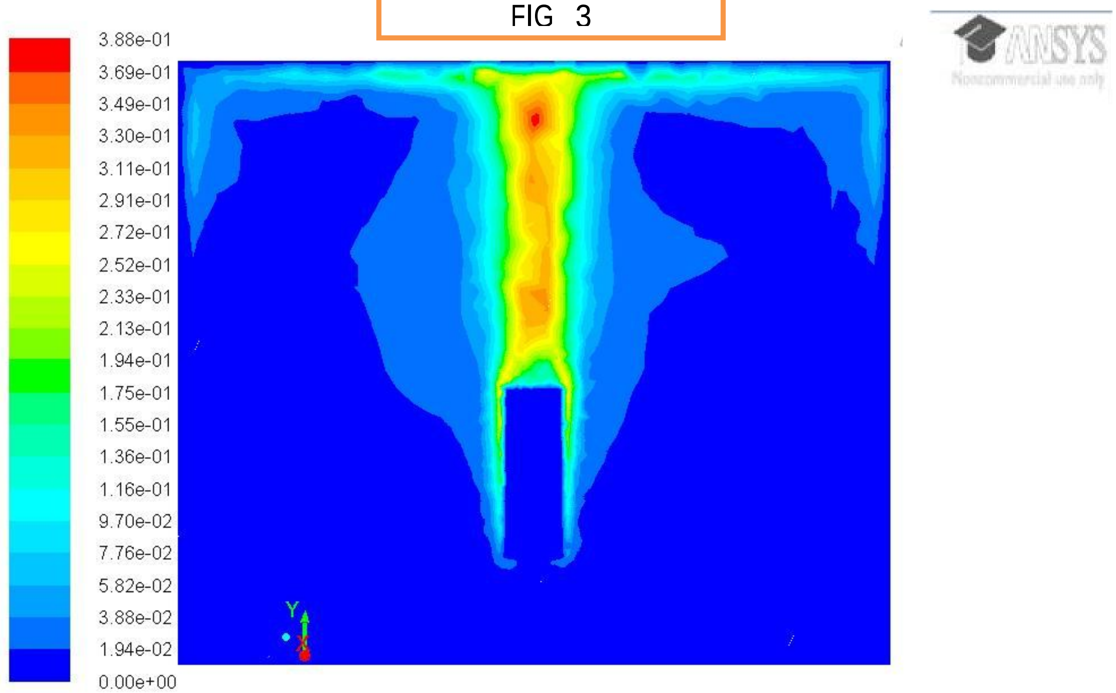


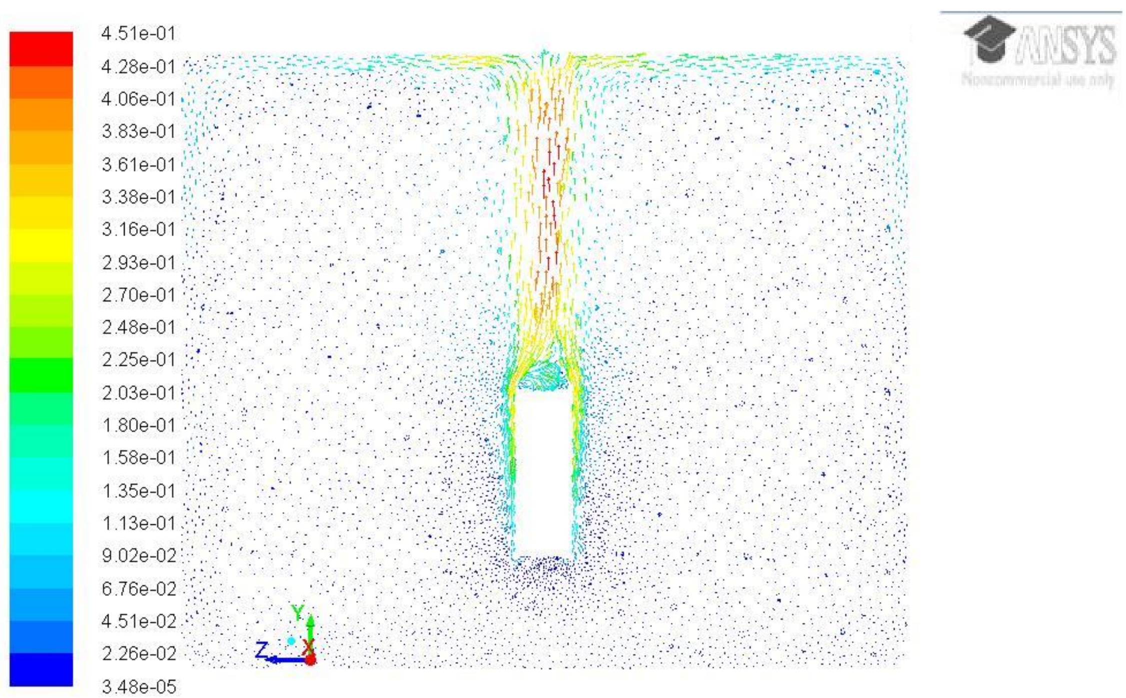
FIG 2

4.1.3 VELOCITY CONTOUR AND VECTOR:

FIG 3



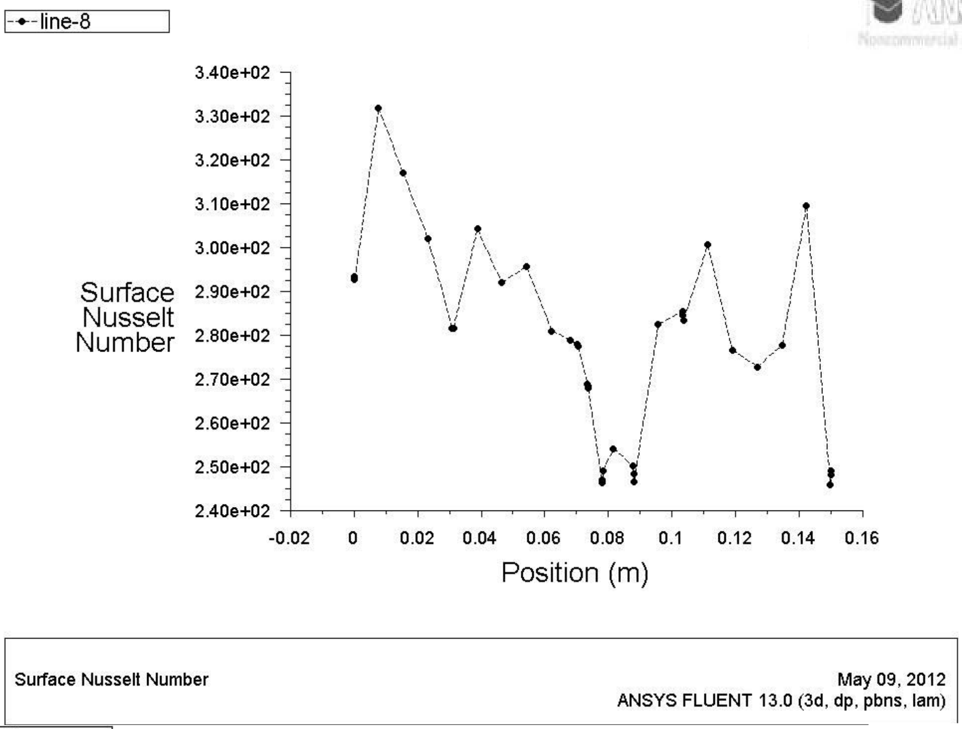
Contours of Velocity Magnitude (m/s) May 09, 2012
ANSYS FLUENT 13.0 (3d, dp, pbns, lam)

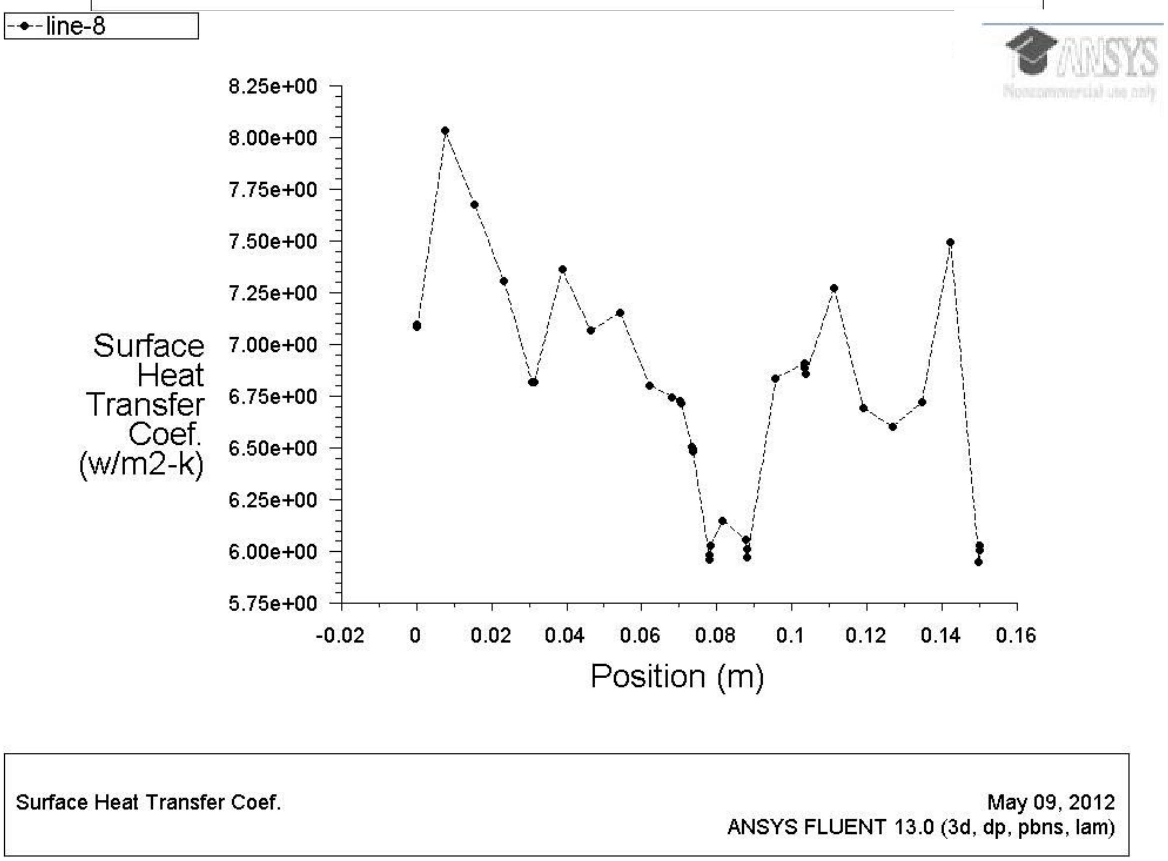


Velocity Vectors Colored By Velocity Magnitude (m/s) May 09, 2012
ANSYS FLUENT 13.0 (3d, dp, pbns, lam)

4.1.4 PLOT FOR NUSSELT NUMBER AND HEAT TRANSFER COEFFICIENT:

FIG 4





4.2 ANALYSIS FOR TUBE WITH CONICAL FINS:

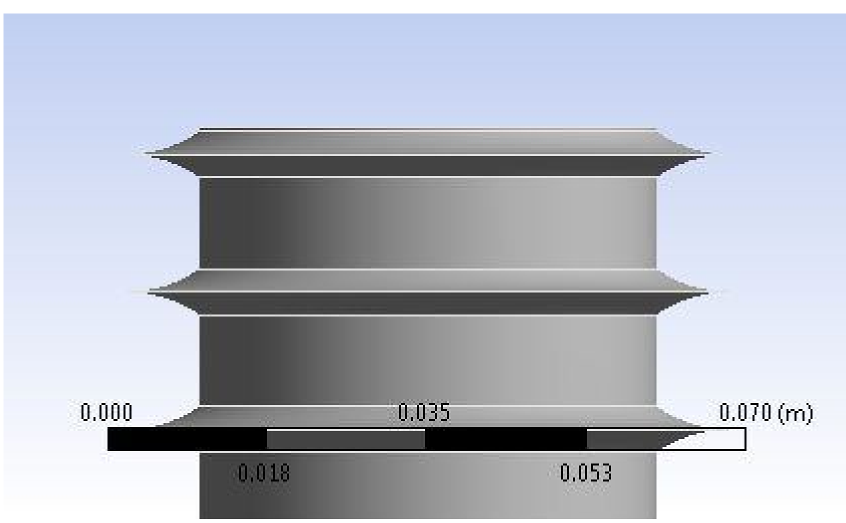
4.2.1 PARAMETERS ASSUMED DURING ANALYSIS:

PHYSICAL PARAMETERS	VALUES
Type of fin	Conical fins base6mmc&width4mm
Cross section of the tube	Circular
External diameter of the pipe	50 mm
Length of the pipe	150 mm
Free stream fluid	Air
Material for tube and fins	Aluminium
Model for convection	Bousinessq
Tube wall temperature	380 k
Free stream air temperature	300 k
Convection heat transfer coff	10 W/m²k
MESH PARAMETERS	VALUES
Messing method	Sweep element
Relevance sizing centre	Fine
Element size	0.0001m
Initial size seed	Active assembly
Smoothing	High
Transition	Slow
Span angle centre	Fine
Number of nodes	2994
Number of elements	2427
Orthogonality quality	3.0812e-01
Aspect ratio	1.82e01

TABLE 2

Cross section of the fin configuration:

FIG 5



4.2.2 TEMPERATURE CONTOUR:

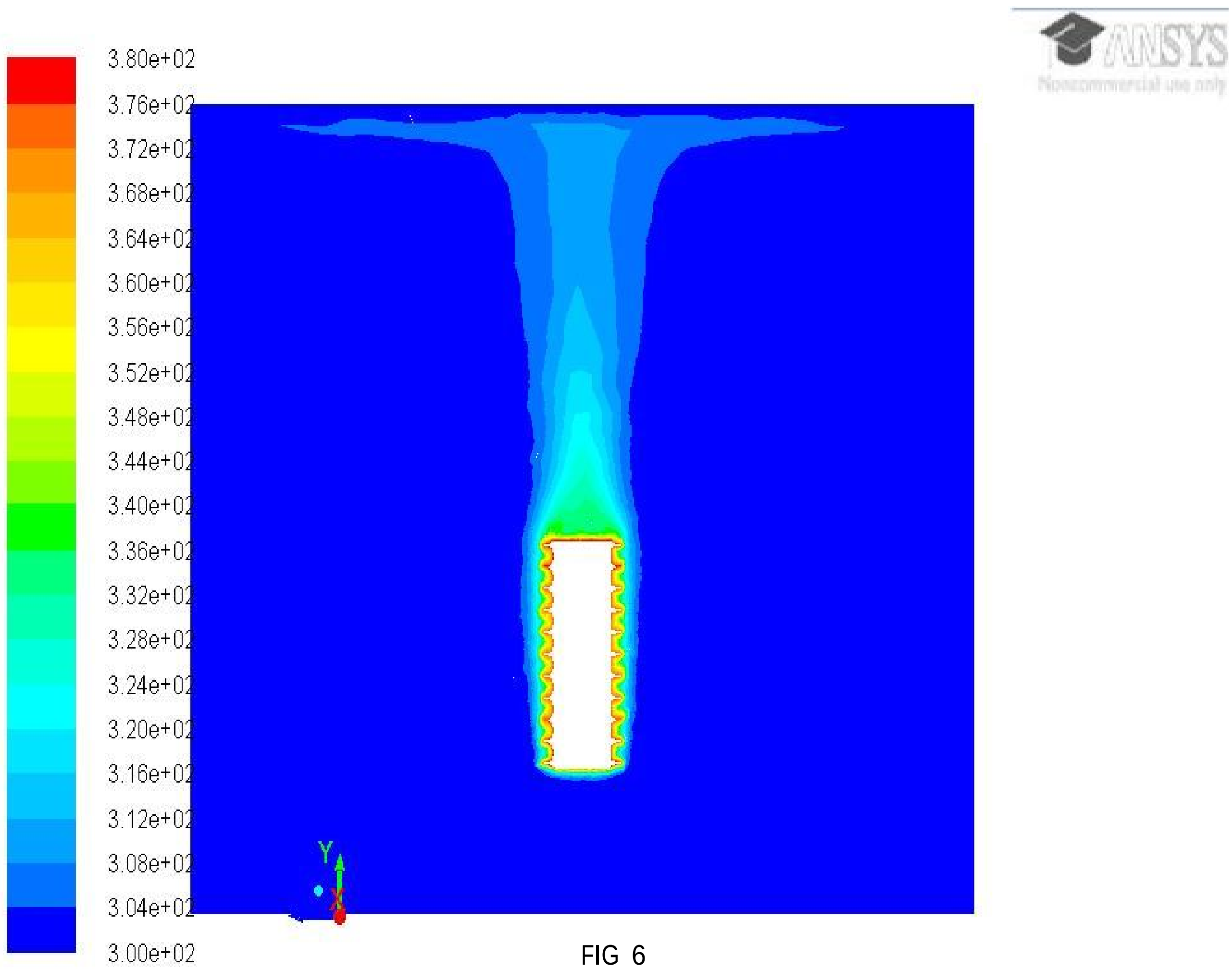


FIG 6

Contours of Static Temperature (k)

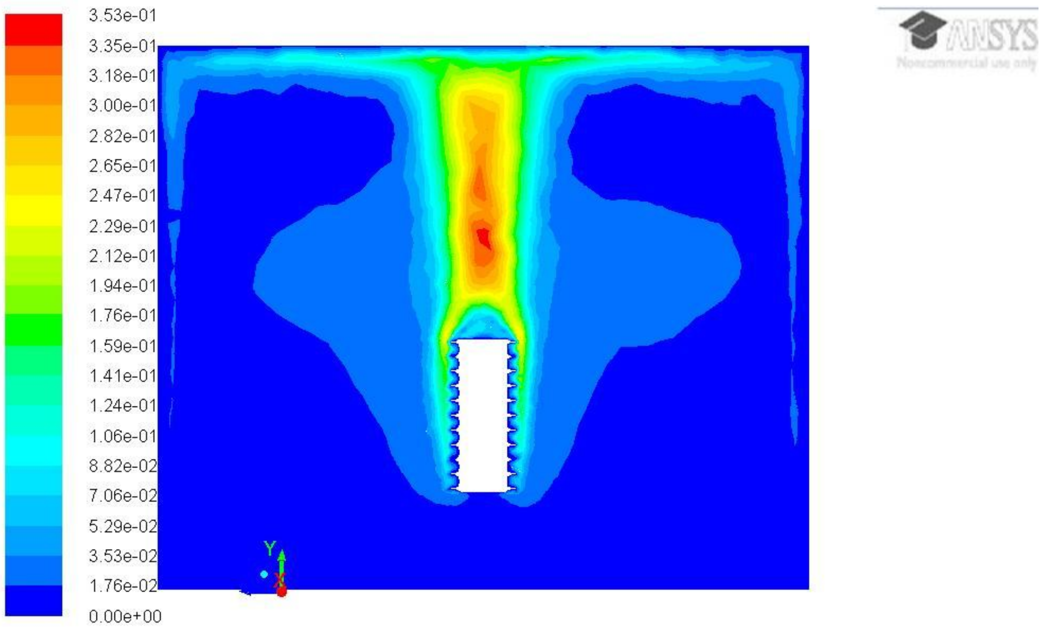
May 09, 2012
ANSYS FLUENT 13.0 (3d, dp, pbns, lam)

MAXIMUM TEMPERATURE ATTAINED: 380 K

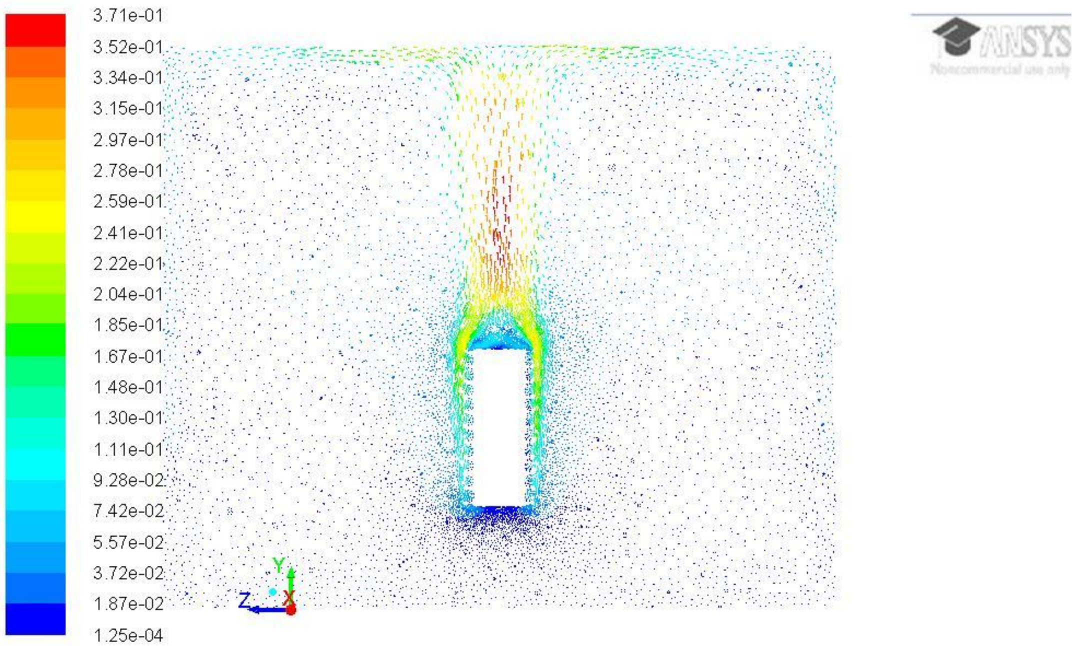
MINIMUM TEMPERATURE ATTAINED: 300 K

CONVECTION LOOPS ARE FORMED AROUND THE ENTIRE PIPE SECTION

4.2.3 VELOCITY CONTOUR:



Contours of Velocity Magnitude (m/s) May 09, 2012
ANSYS FLUENT 13.0 (3d, dp, pbns, lam)

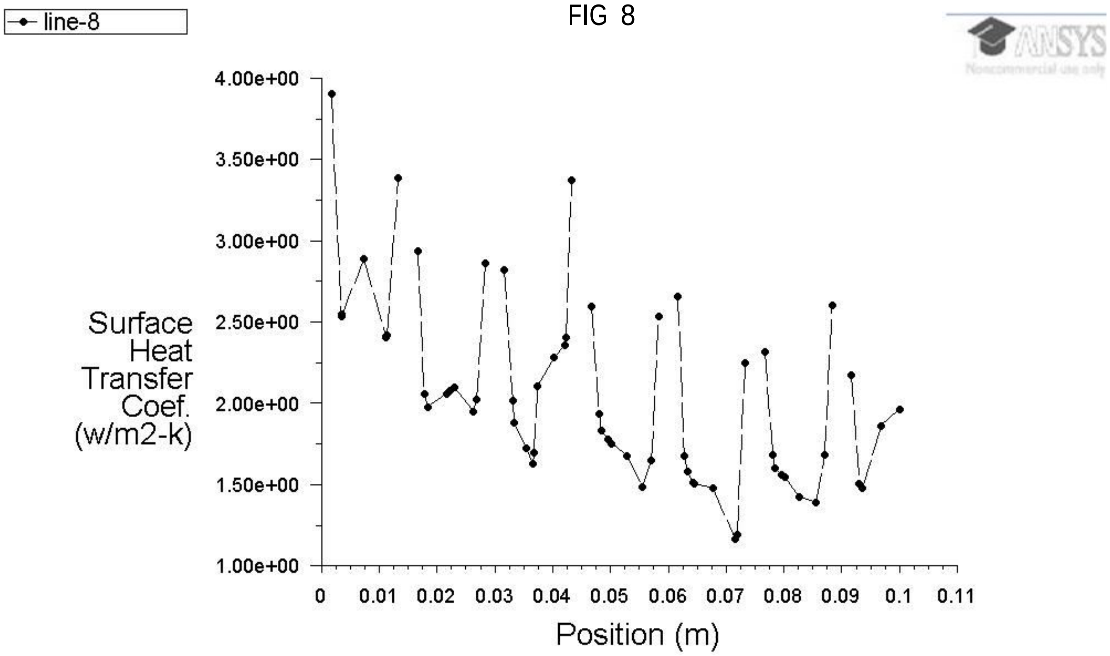


Velocity Vectors Colored By Velocity Magnitude (m/s) May 09, 2012
ANSYS FLUENT 13.0 (3d, dp, pbns, lam)

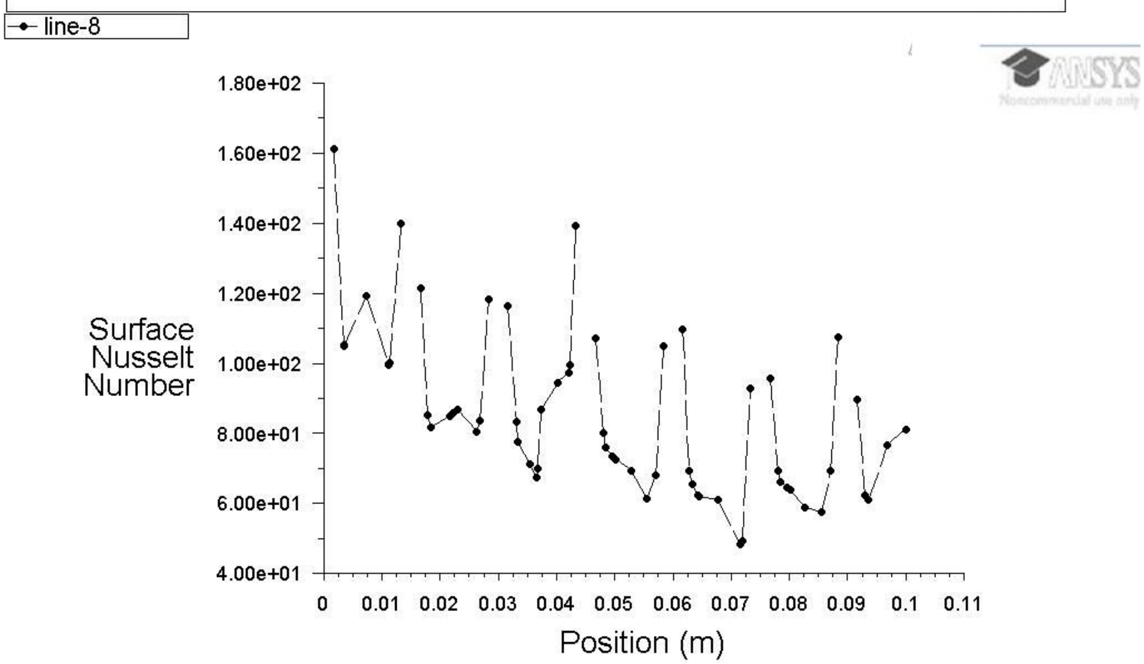
FIG 7

4.2.4 SURFACE NUSSELT NUMBER PLOT:

FIG 8



Surface Heat Transfer Coef. May 09, 2012
ANSYS FLUENT 13.0 (3d, dp, pbns, lam)



Surface Nusselt Number May 09, 2012
ANSYS FLUENT 13.0 (3d, dp, pbns, lam)

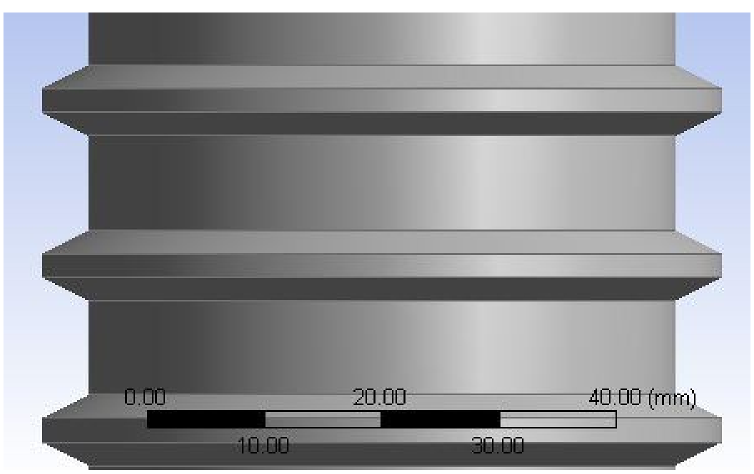
4.3 ANALYSIS FOR TUBE WITH TRAPEZOIDAL FINS

4.2.1 PARAMETERS ASSUMED DURING ANALYSIS:

PHYSICAL PARAMETERS	VALUES
Type of fin	trapezoidal fins base6mm&width4mm
Cross section of the tube	Circular
External diameter of the pipe	50 mm
Length of the pipe	150 mm
Free stream fluid	Air
Material for tube and fins	Aluminium
Model for convection	Bousinessq
Tube wall temperature	380 k
Free stream air temperature	300 k
Convection heat transfer coff	10 W/m²k
MESH PARAMETERS	VALUES
Messing method	Quad/tri elements
Relevance sizing centre	Fine
Element size	0.0001m
Initial size seed	Entire assembly
Smoothing	High
Transition	medium
Span angle centre	Fine
Number of nodes	2834
Number of elements	1954
Orthogonality quality	6.1112e-01
Aspect ratio	1.01232e01

TABLE 3

Cross section of the fin configuration: FIG 9



4.3.1 TEMPERATURE CONTOUR:

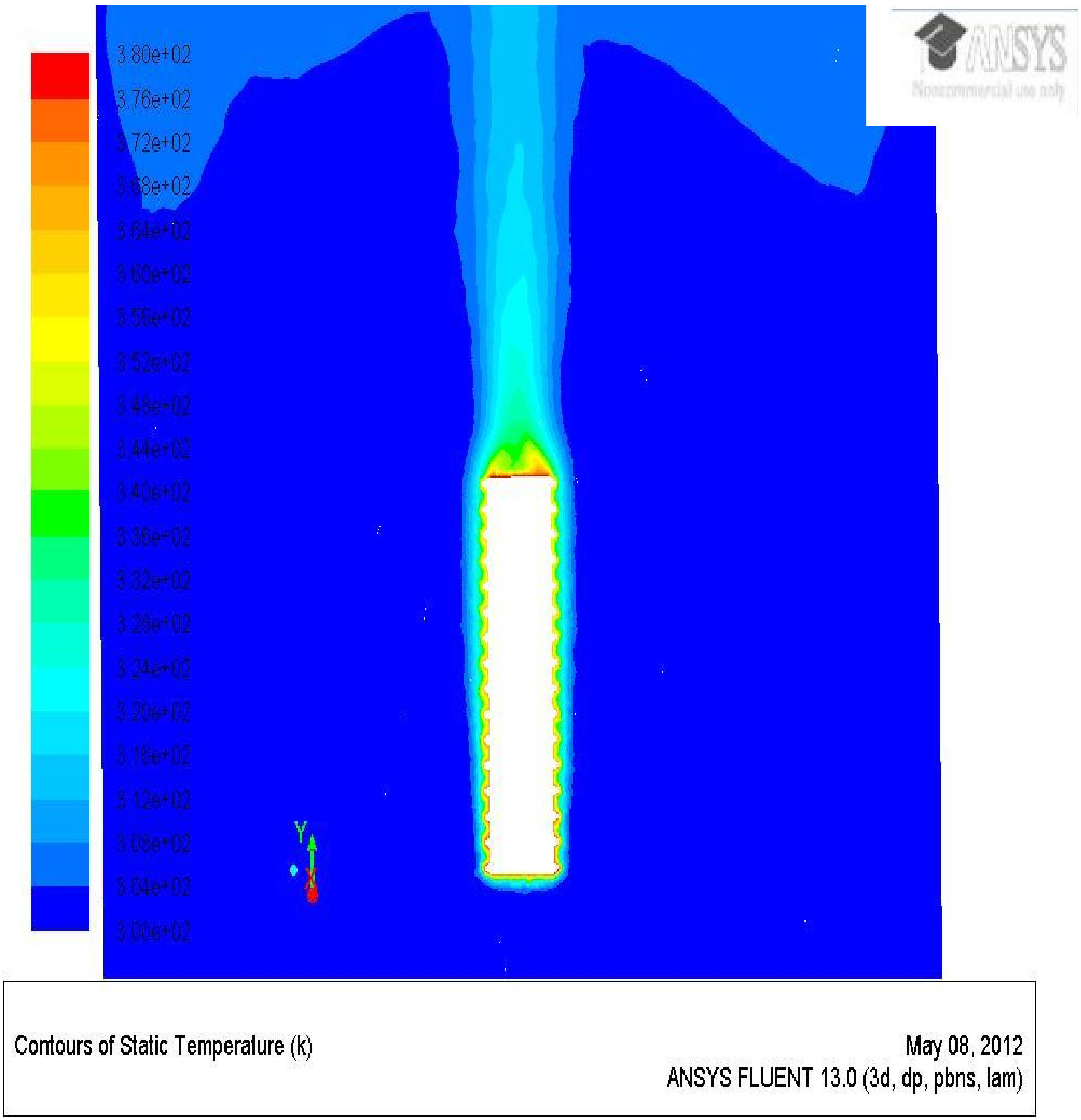
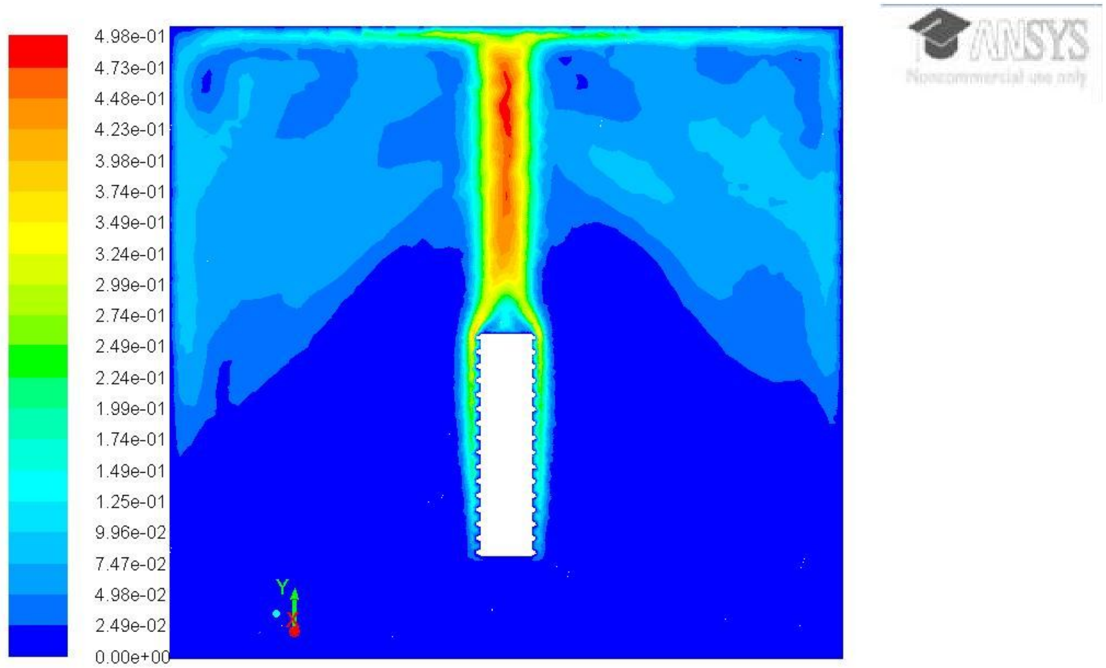
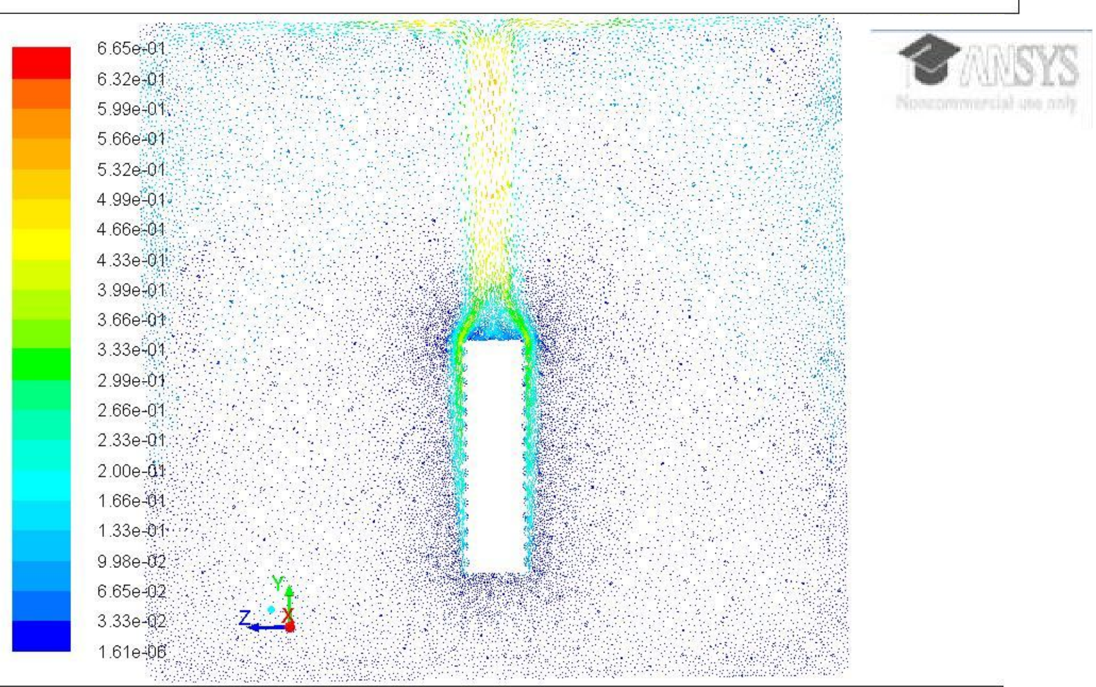


FIG 10

4.3.2 VELOCITY CONTOUR AND VECTORS:



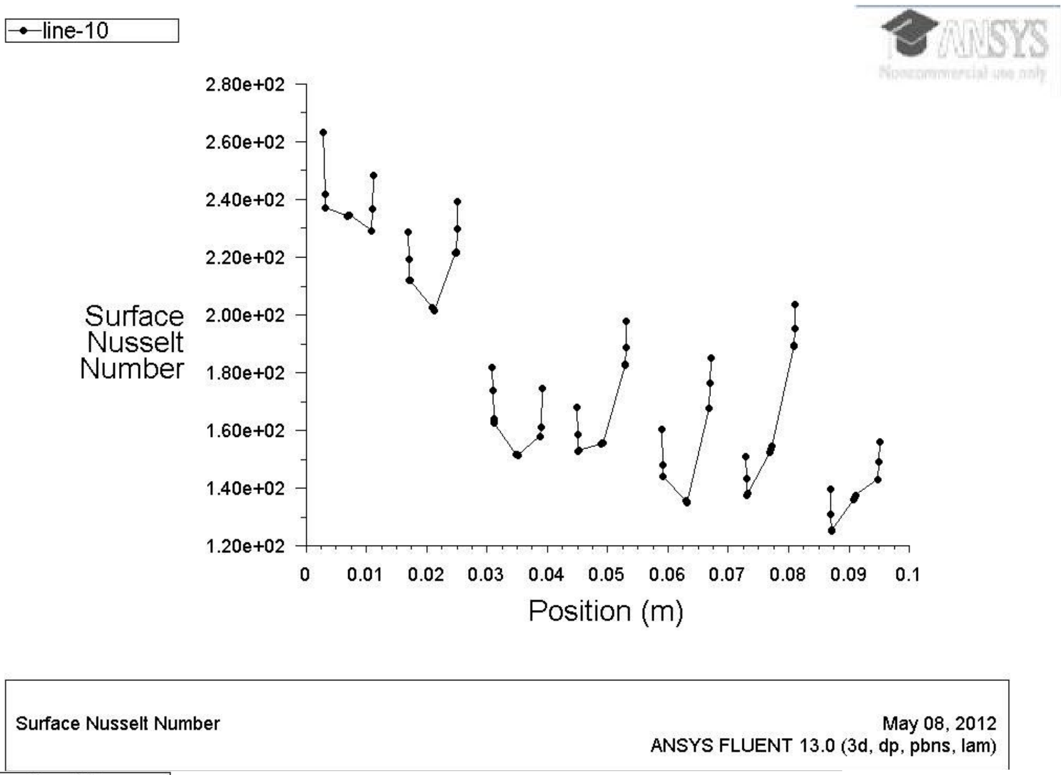
Contours of Velocity Magnitude (m/s) May 09, 2012
ANSYS FLUENT 13.0 (3d, dp, pbns, lam)



Velocity Vectors Colored By Velocity Magnitude (m/s) May 08, 2012
ANSYS FLUENT 13.0 (3d, dp, pbns, lam)

FIG 11

4.3.3 PLOTS FOR DURFACE NUSSELT NUMBER AND HEAT TRANSFER COEFFICIENT:



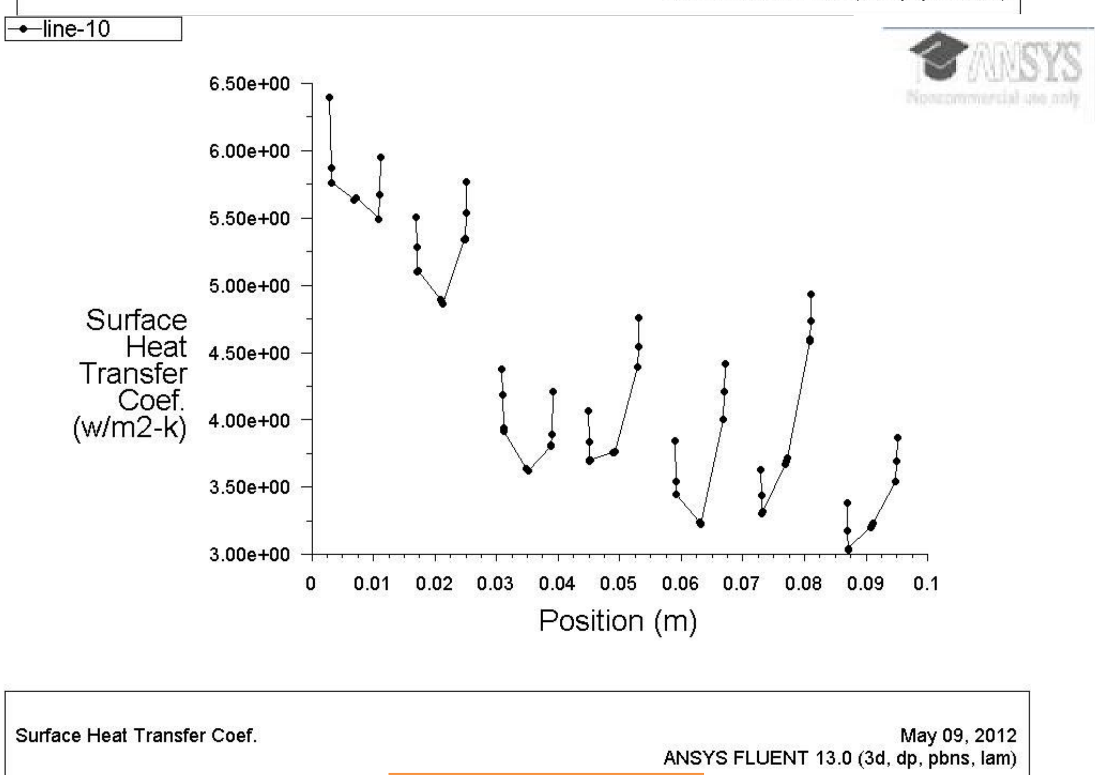


FIG 13

4.4 ANALYSIS FOR TUBE WITH CYLINDRICAL FINS

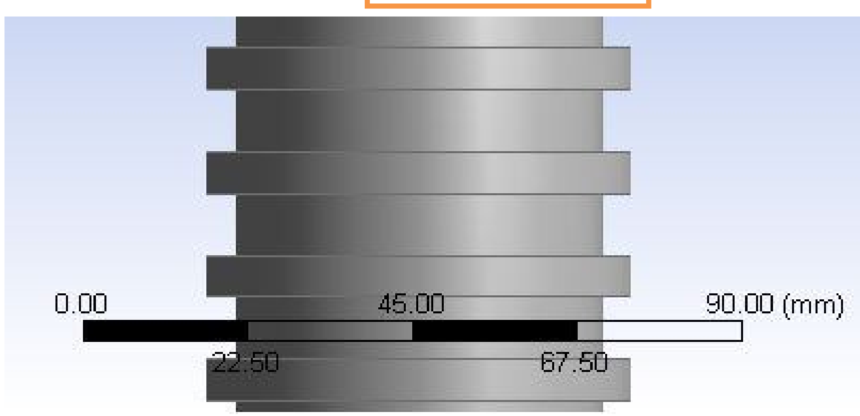
4.4.1 PARAMETERS ASSUMED DURING ANALYSIS:

PHYSICAL PARAMETERS	VALUES
Type of fin	cylindrical fins base6mm&width4mm
Cross section of the tube	Circular
External diameter of the pipe	50 mm
Length of the pipe	150 mm
Free stream fluid	Air
Material for tube and fins	Aluminium
Model for convection	Bousinessq
Tube wall temperature	380 k
Free stream air temperature	300 k
Convection heat transfer coff	10 W/m²k
MESH PARAMETERS	VALUES
Messing method	Quad/tri elements
Relevance sizing centre	Medium
Element size	0.0001m
Initial size seed	active assembly
Smoothing	Medium
Transition	Medium
Span angle centre	Fine
Number of nodes	3256
Number of elements	3054
Orthogonality quality	6.5467e-01
Aspect ratio	2.11282e01

TABLE 4

Cross section of the fin configuration:

FIG 14



4.4.2 TEMPERATURE CONTOUR

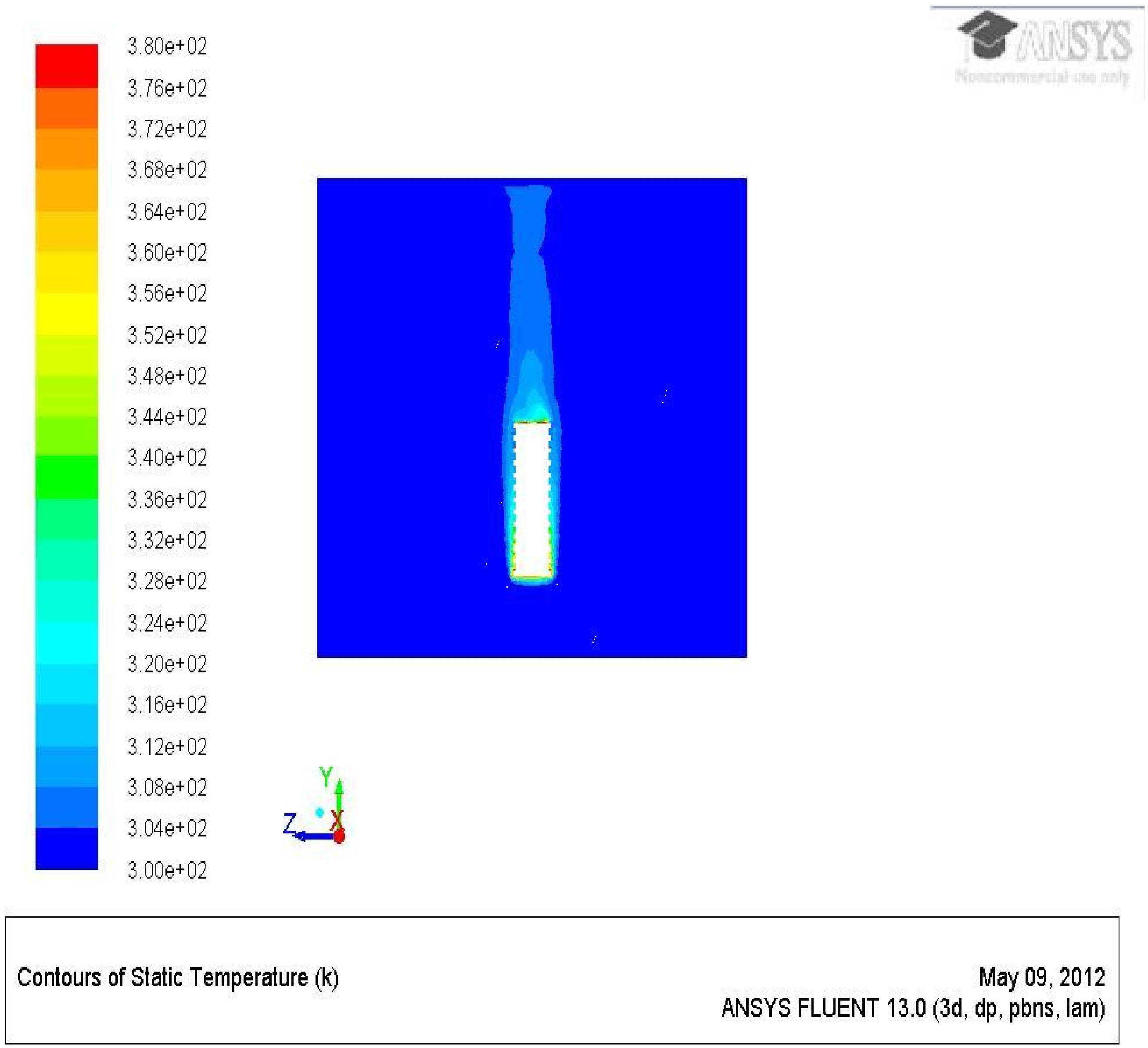
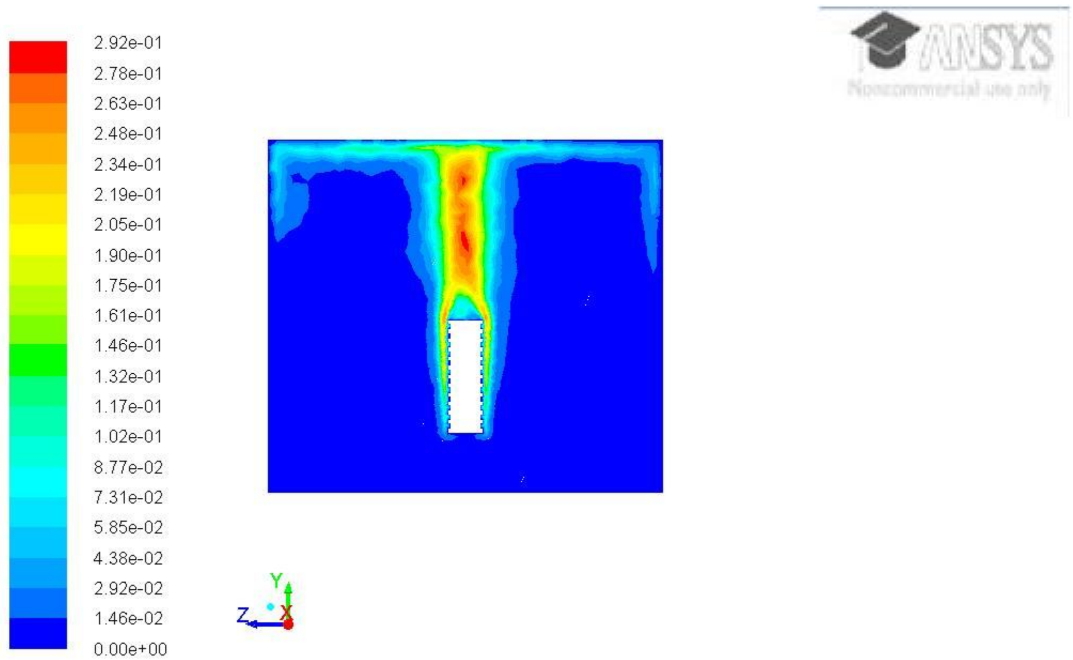
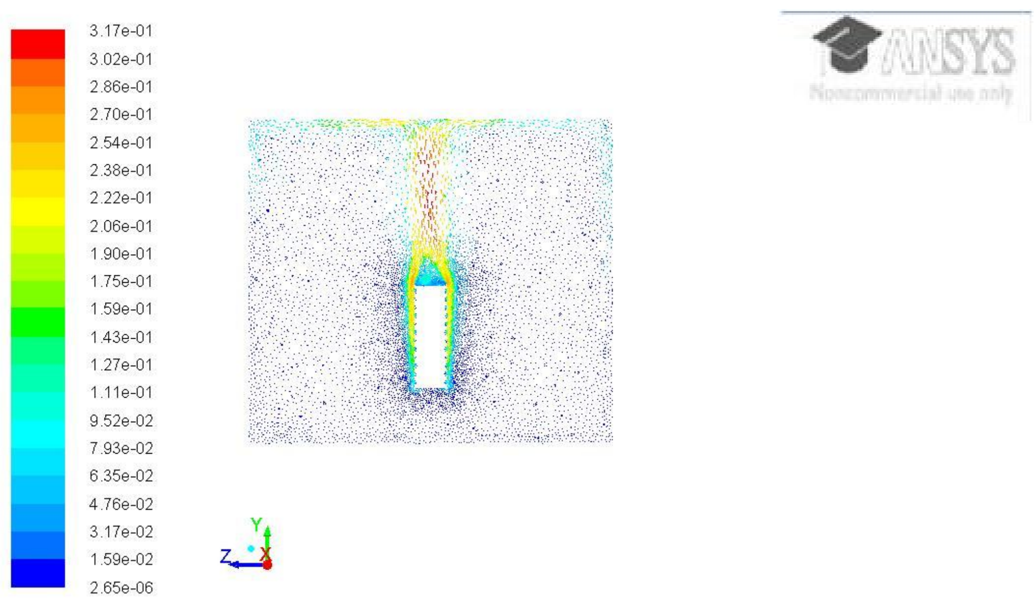


FIG 15

4.4.3 VELOCITY CONTOUR & VECTOR



Contours of Velocity Magnitude (m/s) May 09, 2012
ANSYS FLUENT 13.0 (3d, dp, pbns, lam)

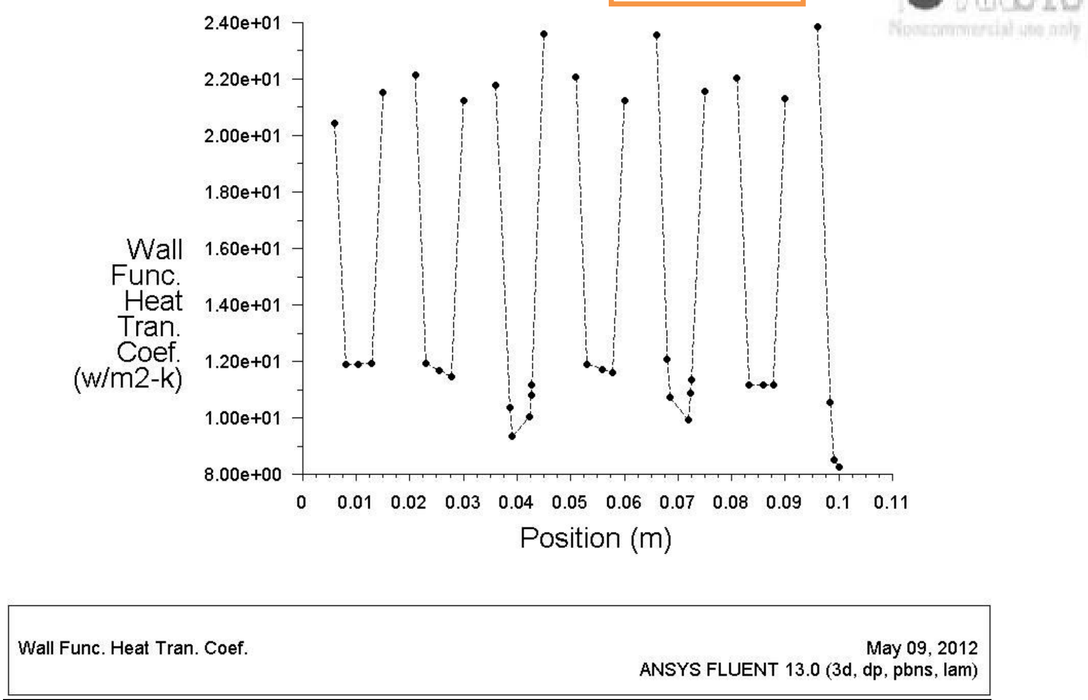


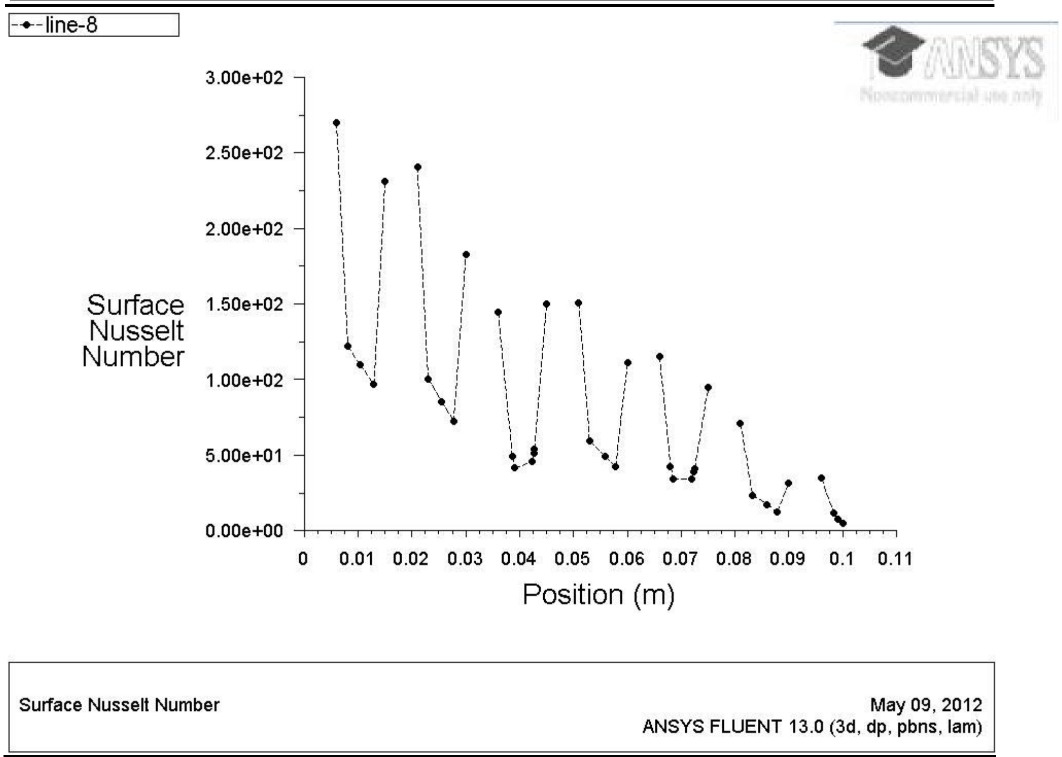
Velocity Vectors Colored By Velocity Magnitude (m/s) May 09, 2012
ANSYS FLUENT 13.0 (3d, dp, pbns, lam)

FIG 16

4.4.4 PLOT OF SURFACE NUSSSELT NUMBER AND HEAT TRANSFER COEFFICIENT

FIG 17





CHAPTER 5

Results and conclusions

5.1 SOLUTION METHODS ADOPTED DURING ANALYSIS IN FLUENT

SCHEME	METHOD
SOLUTION SCHEME:	SIMPLE
GRADIENT:	LEAST SQUARE CELL BASED
PRESSURE:	PRESTO!
MOMENTUM	SECOND ORDER UPWIND
ENERGY:	FIRST ORDER UPWIND

TABLE 5

5.2 BOUSSINESQ METHOD

PARAMETERS	VALUES
DENSITY	1.225 Kg/M^3
SPECIFIC HEAT	1006.43 J/Kg- K
THERMAL CONDUCTIVITY	0.0242 W/m- K
VISCOSITY	1.7894 e -05 kg/ m-S
THERMAL EXPANSION	0.003334 K^{-1}

TABLE 6

5.3 TEMPERATURE CONTOURS

FIG shows the temperature contours for various analyses with various fin configurations. These figures show the variation between the maximum and minimum temperature values across the entire length of the pipe section taken into consideration. Also these contours shows the convection loops formed around the pipe cross section.

MAXIMUM TEMPERATURE = 380 K (near the pipe)

MINIMUM TEMPERATURE = 300 K (ambient air temperature)

5.4 VELOCITY CONTOURS AND VELOCITY VECTORS

FIG shows the velocity contours for various analyses with various fin configurations. These figures show the variation between the maximum and minimum velocity values across the entire length of the pipe section and around the enclosure surrounding the pipe area taken into consideration. Also these contours shows the convection loops formed around the pipe cross section.

PIPE WITHOUT FINS:

MAXIMUM VELOCITY VALUES:	3.88 e -01 mtr/sec
MINIMUM VELOCITY VALUES:	1.94 e -02 mtr/sec

PIPE WITH CONICAL FINS:

MAXIMUM VELOCITY VALUES:	3.53 e -01 mtr/sec
MINIMUM VELOCITY VALUES:	1.76 e -02 mtr/sec

PIPE WITH TRAPEZOIDAL FINS:

MAXIMUM VELOCITY VALUES:	4.98 e -01 mtr/sec
MINIMUM VELOCITY VALUES:	2.49 e 0-02 mtr/sec

PIPE WITH CYLINDRICAL FINS

MAXIMUM VELOCITY VALUES:	2.92 e -01 mtr/sec
MINIMUM VELOCITY VALUES:	1.46 e -02 mtr/sec

5.5 PLOTS FOR SURFACE NUSSELT NUMBER AND SURFACE HEAT TRANSFER CO-EFFICIENT

These plots show the variation of nusselt number across the length of the pipe. The values of the surface nusselt number shows the extent of convective heat loss from the fin surfaces and the outer wall of the heated pipe. Similarly the plot for surface heat transfer coefficient vs the length of the pipe shows the values of surface heat transfer coff. At various points near the outer wall of the pipe and the fin surfaces.

MAXIMUM SURFACE NUSSELT NUMBER AND HEAT TRANSFER COFF:

PIPE WITHOUT FINS: nusselt number = 3.4 e 2 ,
Heat transfer coff. = 4.25 W/m^2-K

PIPE WITH CONICAL FINS: nusselt number = 3.4 e 2 ,
Heat transfer coff. = 4.25 W/m^2-K

PIPE WITH TRAPEZOIDAL FINS: nusselt number = 4.6 e 2 ,
Heat transfer coff. = 6.5 W/m^2-K

PIPE WITH CYLINDRICAL FINS: nusselt number = 3 e 2 ,
Heat transfer coff. = 4.8 W/m^2-K

5.6 HEAT TRANSFER RATE:

PIPE WITHOUT FINS: heat transfer rate at the tube walls = 14.0589 Watts

PIPE WITH CONICAL FINS: heat transfer rate at the tube walls = 3.511642 Watts

heat transfer rate at the tube walls = 11.9164733 Watts

total heat transfer rate = 15.42911 watts

PIPE WITH TRAPEZOIDAL FINS: heat transfer rate at the tube walls = 6.18310 Watts

heat transfer rate at the tube walls = 14.453860 Watts

total heat transfer rate = 20.63697 watts

PIPE WITH TRAPEZOIDAL FINS: heat transfer rate at the tube walls = 4.16301 Watts

heat transfer rate at the tube walls = 11.45617 Watts

total heat transfer rate = 15.61918 watts

5.7 CONCLUSION:

From the above calculated values we can find that the best configuration for this type of convective heat transfer of a heated pipe is a TRAPEZOIDAL fin as they have the highest total heat transfer rate, and the best surface nusselt number along with highest surface heat transfer coefficient.

All the assumptions are made considering the practical manufacturing of fins and the real working conditions. Hence the result thus obtained in the entire project can be referred while dealing with heat transfer related problems where only natural convection is taken into consideration.

5.8 NOMENCLATURE

Heat liberated	Q
Convective heat transfer coeff	h
Heat transfer area	A
temperature of the surface	T_w
ambient air temperature	T_{amb}
free stream temperature	T_∞
temperature	T
density	ρ
acceleration due to gravity	g
velocity	u,v
characteristic length	L
inner radius	r_1
outer radius	r_2
velocity	V
viscosity	μ
constant	β
specific heat capacity	c_p
constant	α

5.9 REFERENCES:

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[8] heat and mass transfer, P k Nag